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Preliminary Test Results Comparing Heat Pump Performance of Refrigerants R-134a and R-12

J.W. Linton, W.K. Snelson, and P.F. Hearty

Technical Report

Rapport technique

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PRELIMINARY TEST RESULTS COMPARING HEAT PUMP PERFORMANCE OF REFRIGERANTS R-134a AND R-12

RÉSULTATS PRÉLIMINAIRES DES ESSAIS DE PERFORMANCE DE RÉFRIGÉRANT R-134a EN COMPARAISON AVEC R-12 DANS UNE POMPE À CHALEUR

J.W. Linton,
W.K. Snelson, and
P.F. Hearty

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I.R.G. Lowe
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Director/
Directeur

ABSTRACT

Performance comparison tests were carried out on R-134a, which is being considered as a substitute refrigerant to replace R-12. The two refrigerants were tested in a small prototype exhaust air heat pump (EAHP) designed for R-12. For the R-134a tests a special compressor lubricating oil was used.

Both refrigerants were tested over the same range of air temperatures supplied to the EAHP. From the test data, graphs were plotted to compare performance of the unit as a function of outdoor air temperature. In this report, comparisons are presented of refrigerant evaporating/condensing temperatures and pressures, heating capacities, compressor power requirements, coefficients of performance (COPs), etc. Thermostatic expansion valve behaviour is also described, including the internal orifice changes which were necessary to provide satisfactory control in the R-134a tests.

RÉSUMÉ

Des essais ont été effectués afin de comparer entre la performance de réfrigérant R-134a et de R-12. Le R-134a étant considéré comme remplacement du R-12. Les essais ont eu lieu dans une petite pompe à chaleur, prototype, à l'air recirculé, construite pour R-12. Un lubrifiant particulier de compresseur a été utilisé avec R-134a.

Les deux fluides frigorigènes ont subi des épreuves à pareil amplitude thermique de l'air amené à la pompe à chaleur. D'après les données les diagrammes ont été tracés pour comparer le comportement de la machine en fonction de température de l'air d'extérieur. Les comparaisons ici comprennent: les températures et pressions d'évaporation et de condensation, les capacités d'échauffement, les puissances absorbées par le compresseur, les coefficients de performance etc. Le comportement du détendeur thermostatique est décrit avec les changements d'orifice nécessaires pour le R-134a.

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PRELIMINARY TEST RESULTS COMPARING HEAT PUMP PERFORMANCE OF REFRIGERANTS R-134a AND R-12

1.0 INTRODUCTION

As a signatory of the Montreal Protocol on Substances that Deplete the Ozone Layer (UNEP 1987), the federal government will impose restrictions in Canada on the consumption of those chlorofluorocarbons (CFCs) designated under the terms of that treaty. Among the fully halogenated CFCs destined for regulation, the refrigerant R-12 is of primary concern. The present total annual consumption of CFCs in Canada amounts to approximately 25 x 10^6 kg, of which at least half is attributed to R-12.

Its favourable thermodynamic properties and suitability for all types of compressors have made R-12 the preferred choice of refrigerant for a wide range of temperature applications. The excellent safety and stability characteristics have also contributed to the versatility of this compound and led to its wide-scale use in the air-conditioning and refrigeration industries. In seeking a long term solution to the replacement of the fully halogenated CFCs, a search is being conducted for alternate new refrigerants which will have comparable properties and at the same time will be environmentally acceptable from an ozone depletion standpoint. A systematic study of CFC refrigerant properties based on the molecular structure (McLinden and Didion 1987) predicted that very few suitable alternatives among the pure fluids would meet all the various constraints. In the case of R-12, the most favourable substitute at the present time appears to be R-134a. Substantial research and development effort is currently being spent on this compound, particularly in terms of toxicology testing and production process technology development (Zurer 1988). It will take several years yet to produce definite toxicity test results, and in the meantime the thermodynamic properties and performance capabilities of R-134a should be fully investigated.

The Low Temperature Laboratory has recently become involved in the testing of new fluids (both pure and mixed refrigerants) as part of a collaboration with DuPont Canada Inc. and Energy, Mines and Resources Canada. One of the first tasks undertaken in this program was a series of small heat pump tests to compare operating performance of the unit using R-12 and R-134a. This report describes the experimental procedure and the corresponding test results obtained.

2.0 TEST EQUIPMENT

2.1 Exhaust Air Heat Pump

The heat pump used in this series of tests was a prototype version of a residential exhaust air heat pump (EAHP) developed and constructed in the laboratory and used previously for proof of concept testing purposes. The EAHP is a heat pump/heat pipe combination proposed for ventilation heat

recovery in tightly constructed R-2000 type homes where some mechanical ventilation is required (Linton 1986). This type of heat pump is intended for indoor installation, with ducted inlet and exhaust air flows drawn through the unit by separate fans. Warm exhaust air passes through one section of the heat pipe and over the evaporator coil (see Figure 1). The condenser is located so that it receives incoming fresh air from the outdoors, and the effect of the heat pipe is to preheat the cool supply air before it reaches the condenser coil. An electric heater is also included upstream of the condenser coil to ensure that the air distributed to the house is maintained above 21.1°C. The refrigerant compressor used is a small hermetically sealed reciprocating type. The particular system tested was a smaller version of the EAHP described in detail elsewhere (Linton 1988).

The EAHP was considered to be a suitable test device for this purpose, since it was designed to operate with refrigerant R-12 and owing to its small size it only required a minimal refrigerant charge. The latter consideration was important since the quantity of R-134a available was very limited.

2.2 Psychrometric Calorimeter

For testing purposes the EAHP was located in a psychrometric calorimeter. The calorimeter is a facility designed and constructed specifically for the purpose of testing heat pump and air-conditioning equipment. It consists basically of two environmental chambers capable of reproducing a wide range of indoor and outdoor test conditions. Extensive instrumentation and control equipment maintain precise operating conditions within the chambers, and the computer-controlled data acquisition system takes readings as required.

2.3 Test Unit Installation

The EAHP was installed within the indoor chamber with ducted connections as shown in Figure 2. Exhaust air was drawn from the indoor room through the EAHP and then through an airflow meter and supplementary fan to the outdoor room. The outdoor test chamber was used as the source for conditioned fresh air. The supply air was drawn through the EAHP and then through an airflow meter and supplementary fan into the indoor test chamber.

3.0 TEST PROCEDURE

3.1 Airflow Test Measurements

The indoor air conditions were kept constant throughout the tests at 22°C and 40% relative humidity. Outdoor ambient temperatures were varied between -20 and 20°C. Airflow rates for the supply and exhaust streams were measured using ASME standard long radius flow nozzles located in the ducting. Dampers were used to adjust pressure drops and maintain balanced flow rates of 70.8 standard L/s on each side of the unit.

The inlet and outlet temperatures of the supply and exhaust air streams were measured by thermocouple grids located in the air ducts. Overall temperature

differences of each airstream were measured using thermopiles. Other air temperatures, dew points, and pressures were measured at points elsewhere as indicated in Figure 3.

3.2 Refrigerant Test Measurements

The refrigerant circuit was set up to provide typical operating conditions for evaporator outlet superheat and condenser outlet subcooling. For both refrigerants the evaporator superheat was set at 5.6°C at an outdoor air temperature of 15°C, and the charge in each case was adjusted in an attempt to provide approximately the same level of condenser subcooling. Refrigerant temperatures and pressures were measured at suitable points in the circuit as shown in Figure 4.

3.3 Refrigerant Changeout

On completion of the baseline tests with R-12, the compressor and system were drained, flushed with R-11, and recharged with R-134a according to a recommended procedure supplied by DuPont Canada Inc. The alkylbenzene-based lubricating oil was replaced by a special polyalkylene-glycol based synthetic oil supplied for this purpose by DuPont. The same series of tests were then repeated for R-134a.

4.0 TEST RESULTS

A summary of the test data obtained is shown in Table 1. Results were extracted from the test data and plotted in a series of graphs comparing the behavior of some important parameters as a function of outdoor air temperature. However, at lower temperatures the EAHP requires the electric heater to come on, which produces discontinuities in readings for that region. Therefore, for reasons of clarity, the results obtained for outdoor temperatures below -5°C have been omitted from the graphs presented in this report.

4.1 Evaporating/Condensing and Compressor Discharge Conditions

Measured values of evaporating and condensing pressures for both refrigerants are shown in Figure 5. Evaporating conditions are almost identical, but condensing pressures are consistently higher for R-134a by nearly 138 kPa. These pressures correspond to evaporating and condensing temperatures that were derived using saturation properties data for both fluids (see Figure 6). The pressure/temperature relations for R-134a were taken from data by Wilson and Basu (1988). It can be seen that the evaporating and condensing temperatures for both refrigerants are in close agreement over the whole temperature range, with some slight divergence of condensing temperatures occurring at lower outdoor temperatures. Measured values of compressor discharge temperatures are also included in Figure 6, with the temperature readings being 5.6 to 11.1°C higher for R-134a than for R-12 owing to the higher condensing pressure condition in the case of R-134a. Corresponding compressor discharge superheats are plotted in Figure 7.

4.2 Heating Capacity

Heating capacities are compared in Figure 8. It should be noted that the capacity of the EAHP increases as the outdoor temperature falls, mainly owing to the presence of the heat pipe (the larger the temperature gradient between the supply and exhaust air, the greater the capacity of the heat pipe). The capacity of the heat pump also increases because of the reduction of the condensing temperature. There is close agreement between heating capacities for the two refrigerants, with slightly higher values indicated for R-134a at lower outdoor temperatures. This small amount of divergence in heating capacity is consistent with the condensing temperature effect observed in Figure 6.

4.3 Compression Ratios and Power Inputs

Compression ratios were derived directly from the evaporating and condensing pressures of Figure 5, and the corresponding values are plotted in Figure 9. This shows that the compression ratio is 5 to 10% higher for R-134a over the range of outdoor test temperatures. These higher compression ratios can be related to the required compressor power and Figure 10 shows that the measured compressor electrical energy input required for R-134a is about 20% higher than for R-12.

4.4 Coefficient of Performance

Based on the measured values of heating capacity and compressor power, the corresponding coefficients of performance (COPs) are approximately 15% less for R-134a (see Figure 11). The trend to higher COPs at lower outdoor temperatures follows the heating capacity characteristic behaviour of the EAHP described above.

4.5 Expansion Valve Performance

The thermostatic expansion valve used in this series of tests was an externally equalized unit selected for refrigerant R-12 and sized according to operating conditions required by the wide range of EAHP supply air temperatures. The valve had interchangeable orifices and the one chosen had a nominal capacity of 1.05 kW, based on 4.4°C evaporating and 32.2°C condensing temperatures with 5.5°C subcooling.

In the case of R-12 the valve controlled well over the whole test range, as seen in Figure 12. The evaporator superheat varied by only about 1.1°C from its set point of 5.6°C and the subcooling was also fairly constant. However, using the same orifice (orifice #1) for R-134a produced very high levels of superheat and the valve was clearly out of its control range. This behaviour is illustrated in Table 2, which shows the superheat and subcooling measurements for two typical operating conditions.

In an attempt to overcome this problem, the next size larger orifice (orifice #2, corresponding to 1.76 kW nominal rating) was installed and the R-134a

tests repeated (see Table 2). It can be seen again from the erratic superheat behaviour for both operating conditions that the valve was not controlling as required.

Orifice #3 was a 3.50 kW nominally rated unit. With this orifice in place satisfactory superheat readings were obtained, as seen for the corresponding two conditions in Table 2. Figure 12 also shows that using this orifice the superheat and subcooling for R-134a followed closely the corresponding conditions for R-12 throughout the temperature range, indicating that the valve was controlling well.

5.0 DISCUSSION

The higher compression ratios observed for R-134a [Figure 9] indicate lower volumetric efficiencies for the compressor. This means that the actual volumes of suction vapour compressed per unit time will be less for R-134a, and because the heat content of the vapour per unit volume is approximately the same for both refrigerants, then it follows that the refrigerating (evaporating) capacity will be less for R-134a. Lower evaporating capacities are also consistent with the results shown in Figures 8 and 10. That is,for the same heating capacity and higher compressor power requirements, it is expected that the corresponding evaporating capacities of R-134a would be lower.

The performance of the expansion valve is difficult to explain. Calculations of refrigerant flows based on heating capacities and enthalpy changes indicate that substantially lower flows would be expected for R-134a for a given operating condition. Therefore, one would anticipate that a smaller orifice would be required in the valve rather than a larger one. This controversial valve behaviour is still unanswered at this time, and further investigation work is planned.

6.0 CONCLUSIONS

Based on the results obtained for performance comparison tests of R-12 and R-134a in the prototype EAHP, it was shown that approximately the same heating (output) capacity was achieved for both refrigerants over the required range of outdoor air supply temperatures.

The evaporating pressure and the evaporating and condensing temperatures were also about the same, but the condensing pressures were about 138 kPa higher for R-134a. The effect of the higher pressures was reflected in higher compressor power requirements and lower COPs. On average, the COP for R-134a was 15% less than that for R-12 over the outdoor temperature range.

For R-134a tests,a much larger orifice was required for satisfactory control of the R-12 thermostatic expansion valve. The corresponding nominal rated capacity of the valve was over three times greater than that of the original installation.

No equipment problems were experienced during 1300 h of continuous operation of the EAHP using R-134a and the polyethylene glycol oil.

For applications such as the EAHP, refrigerant R-134a can clearly be considered as an alternative refrigerant. However, for practical purposes, it is unlikely to be used in retrofit situations, owing to the draining/flushing/recharging difficulties and expansion valve problems involved. In the case of new equipment, the system can be redesigned to avoid such problems and to make other accommodations if required. For example, if a COP equivalent to the R-12 system is necessary, then the new system can be designed with additional condensing surface area (at correspondingly higher cost) to reduce condenser pressures.

7.0 REFERENCES

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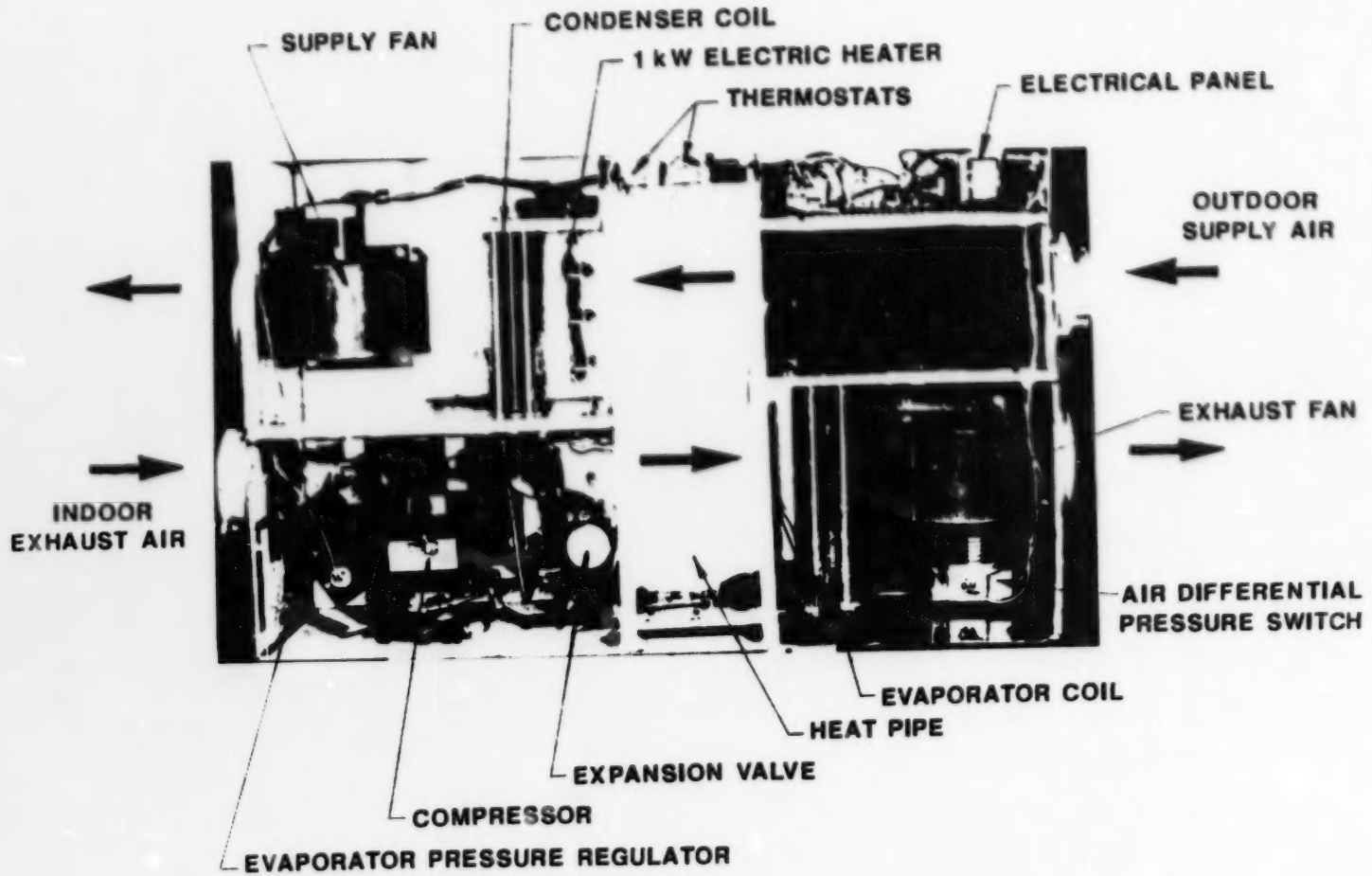

Figure 1. Exhaust air heat pump - plan view

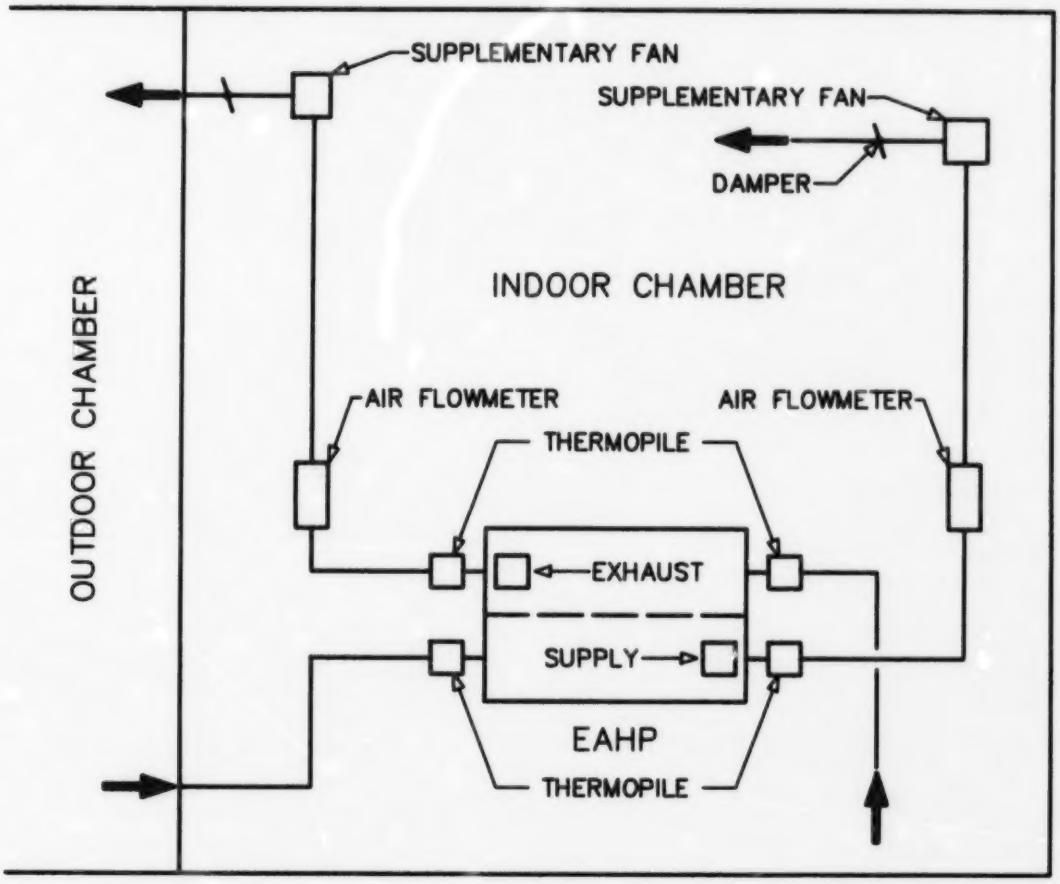

Figure 2. EAHP air flow test schematic in psychrometric calorimeter

TEST NO.	____	
DATE	____	
REFRIGERANT	____	
EXHAUST AIR FLOW	____	L/s
EXHAUST AIR FAN POWER	____	W
FRESH AIR FLOW	____	L/s
FRESH AIR FAN POWER	____	W
ΔT ACROSS FRESH AIR SIDE	____	°C
HEAT SUPPLIED TO FRESH AIR (HEATING CAPACITY)	____	W
HEATER POWER	____	W
COMPRESSOR POWER	____	W
COP (INCLUDING FAN POWER)	____	

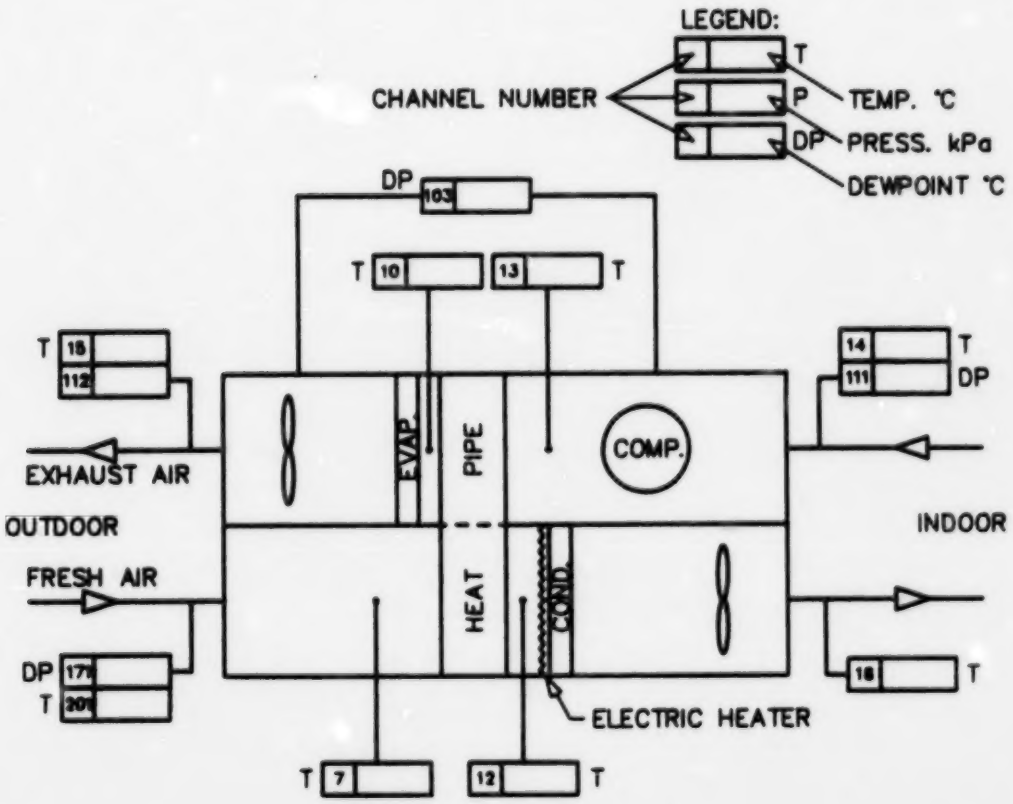

Figure 3 Exhaust air heat pump – air flow test measurement data record sheet

TEST NO. ____
DATE ____
REFRIGERANT ____

EVAPORATOR SUPERHEAT ____ °C
CONDENSER SUBCOOLING ____ °C
COMPRESSION RATIO ____

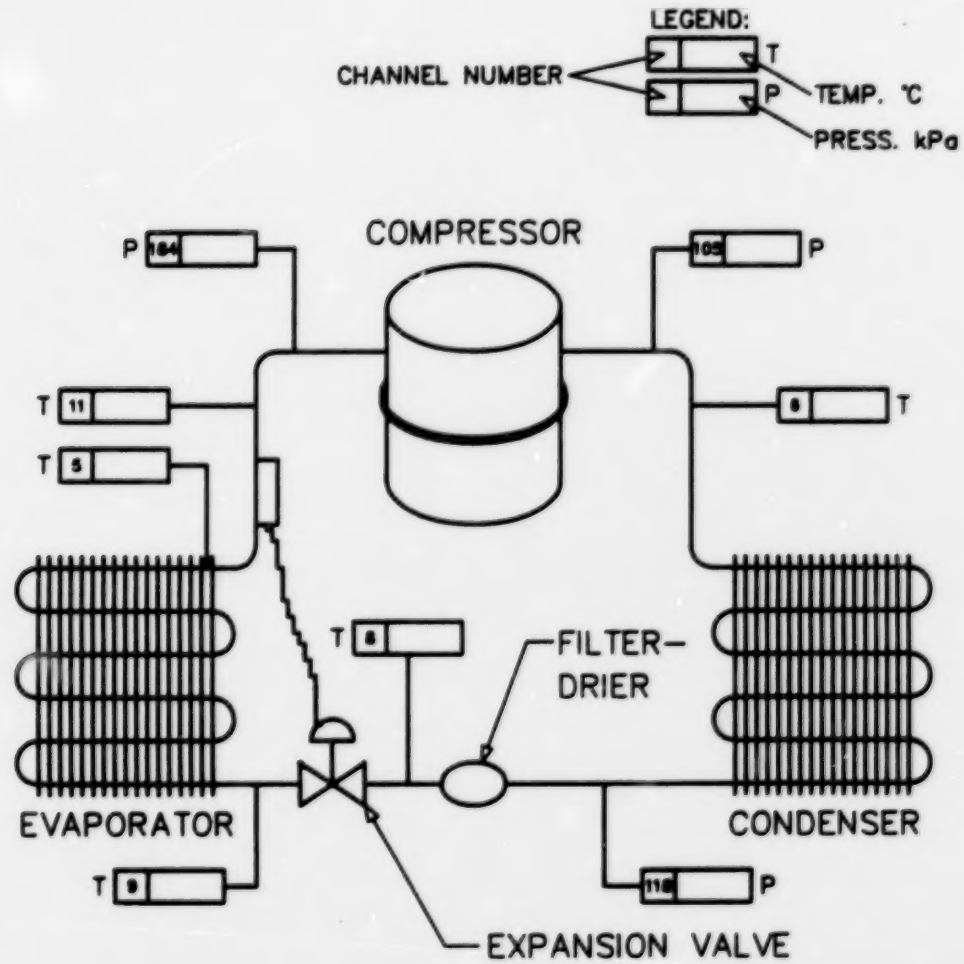

Figure 4 Exhaust air heat pump – refrigerant circuit test measurement data record sheet

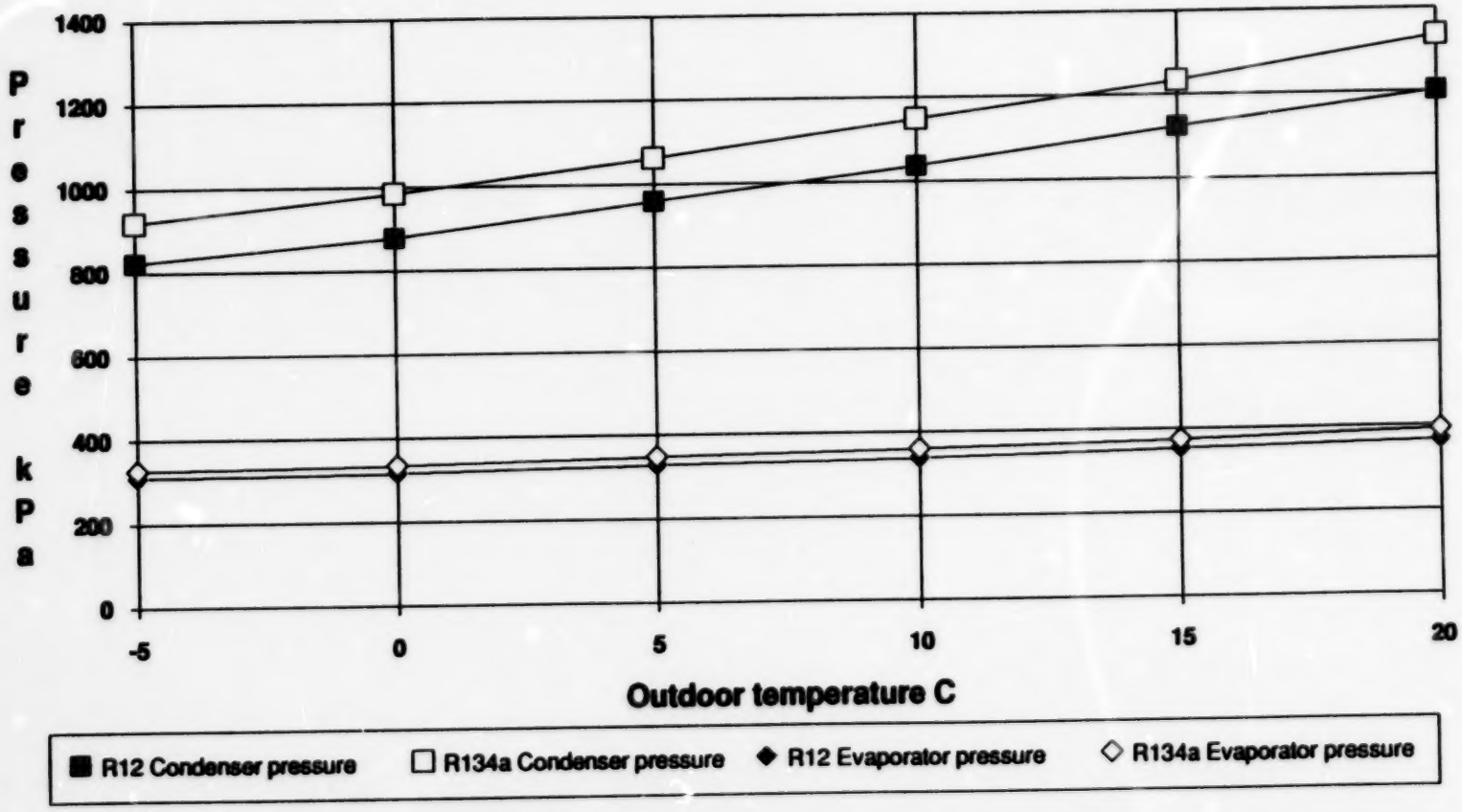

Figure 5. Comparison of R12 and R134a evaporating and condensing pressure

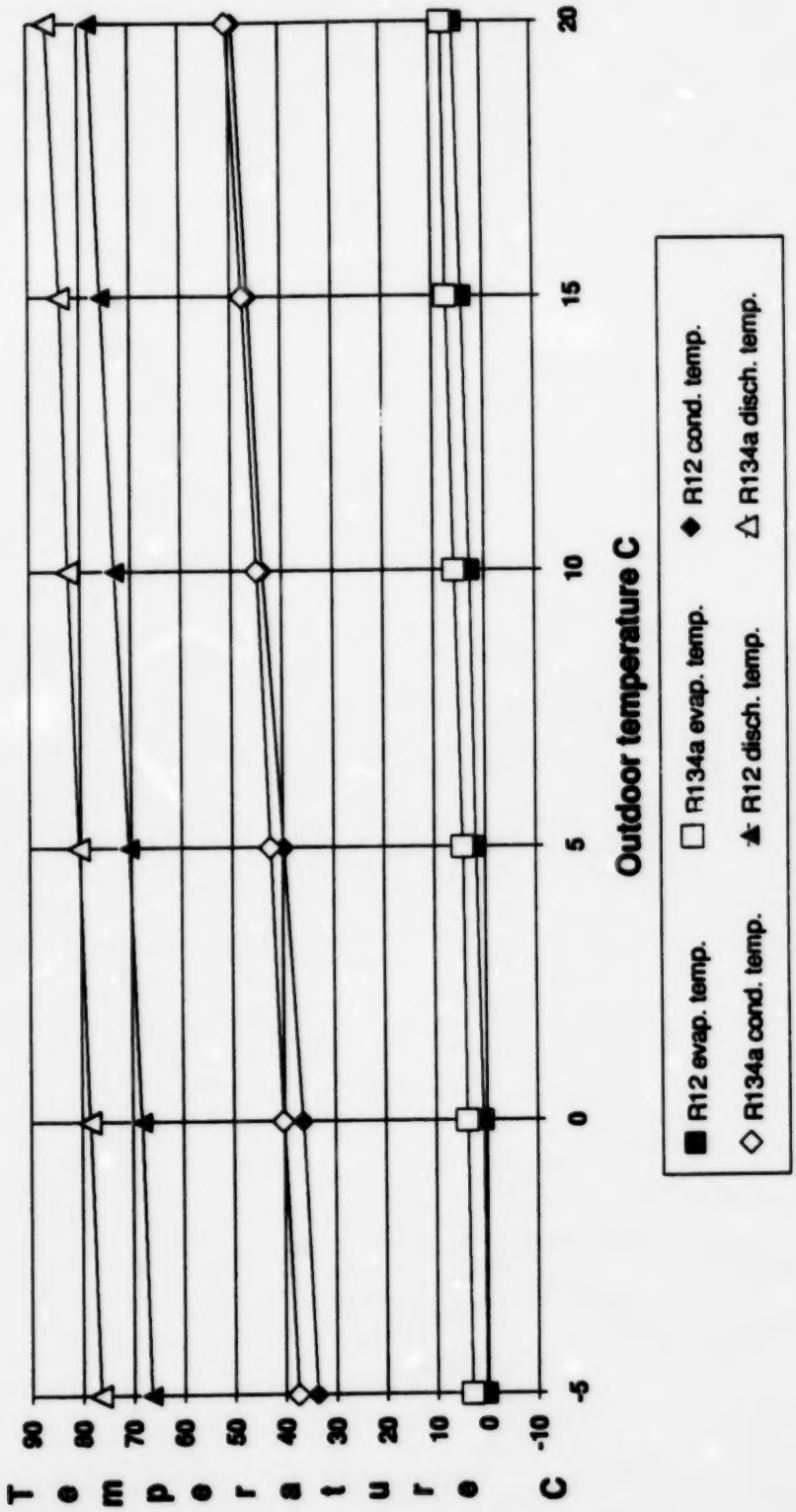

Figure 6. Comparison of R12 and R134a evaporating, condensing, and compressor discharge temperature

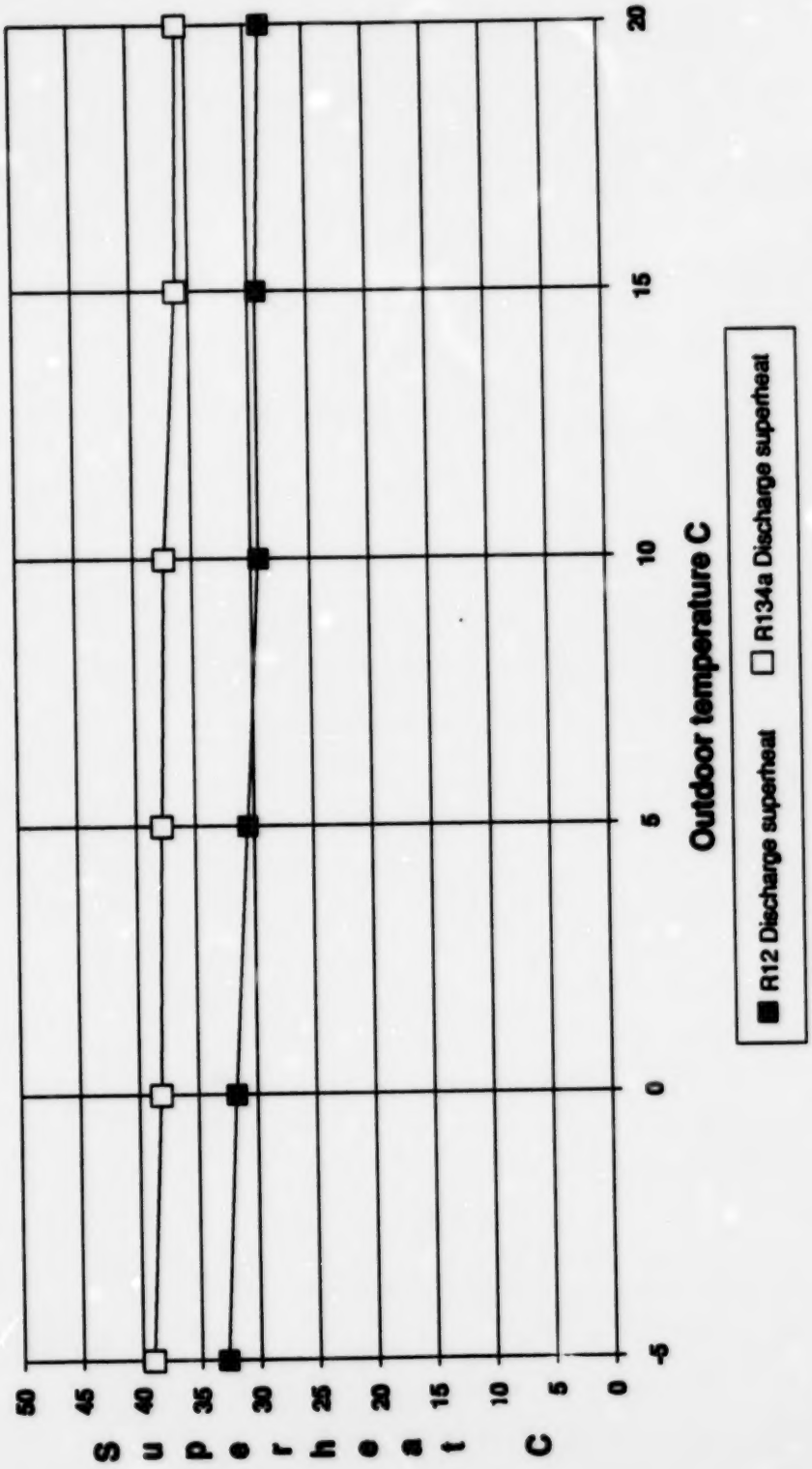

Figure 7. Comparison of R12 and R134a compressor discharge superheat

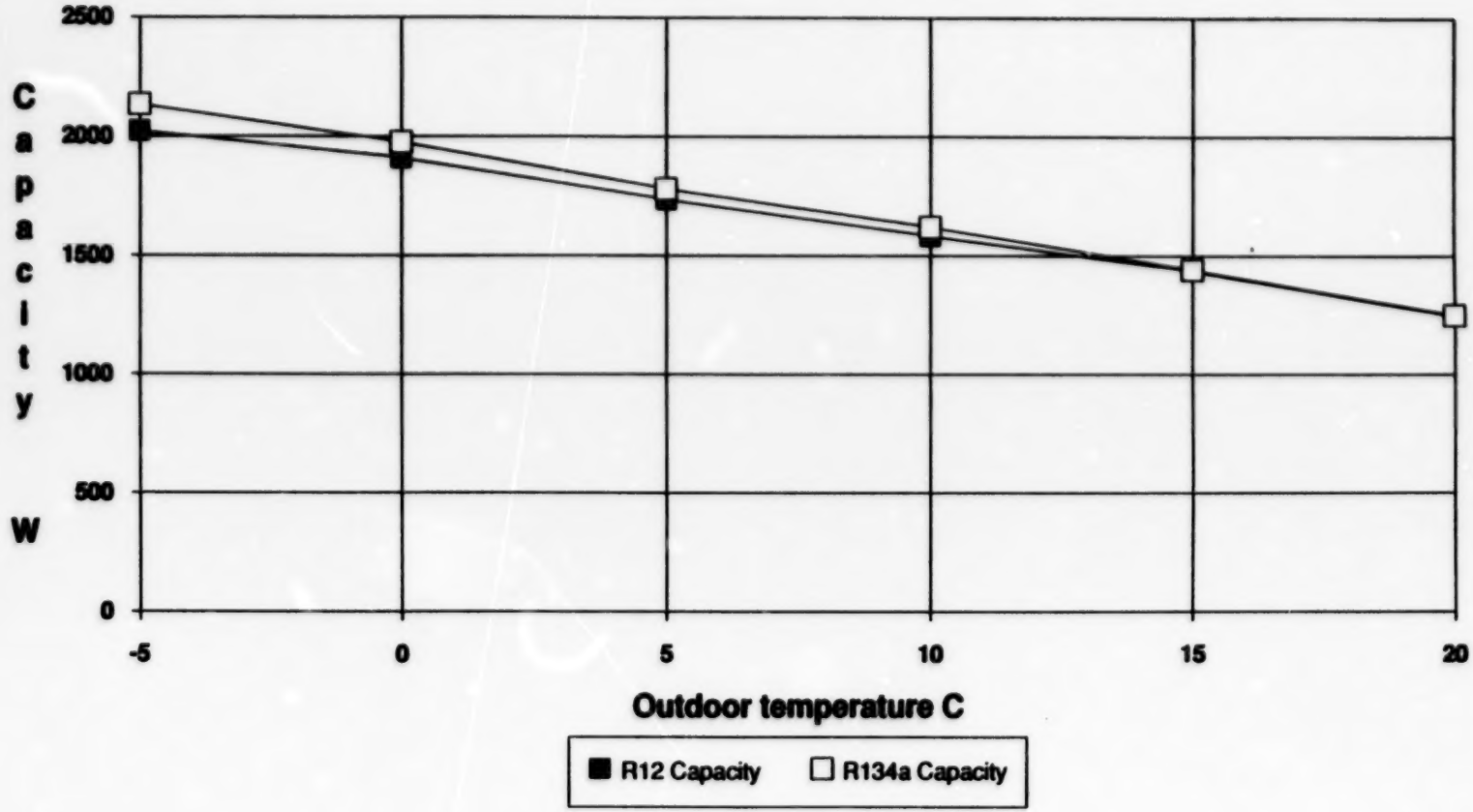

Figure 8. Comparison of R12 and R134a heating capacity

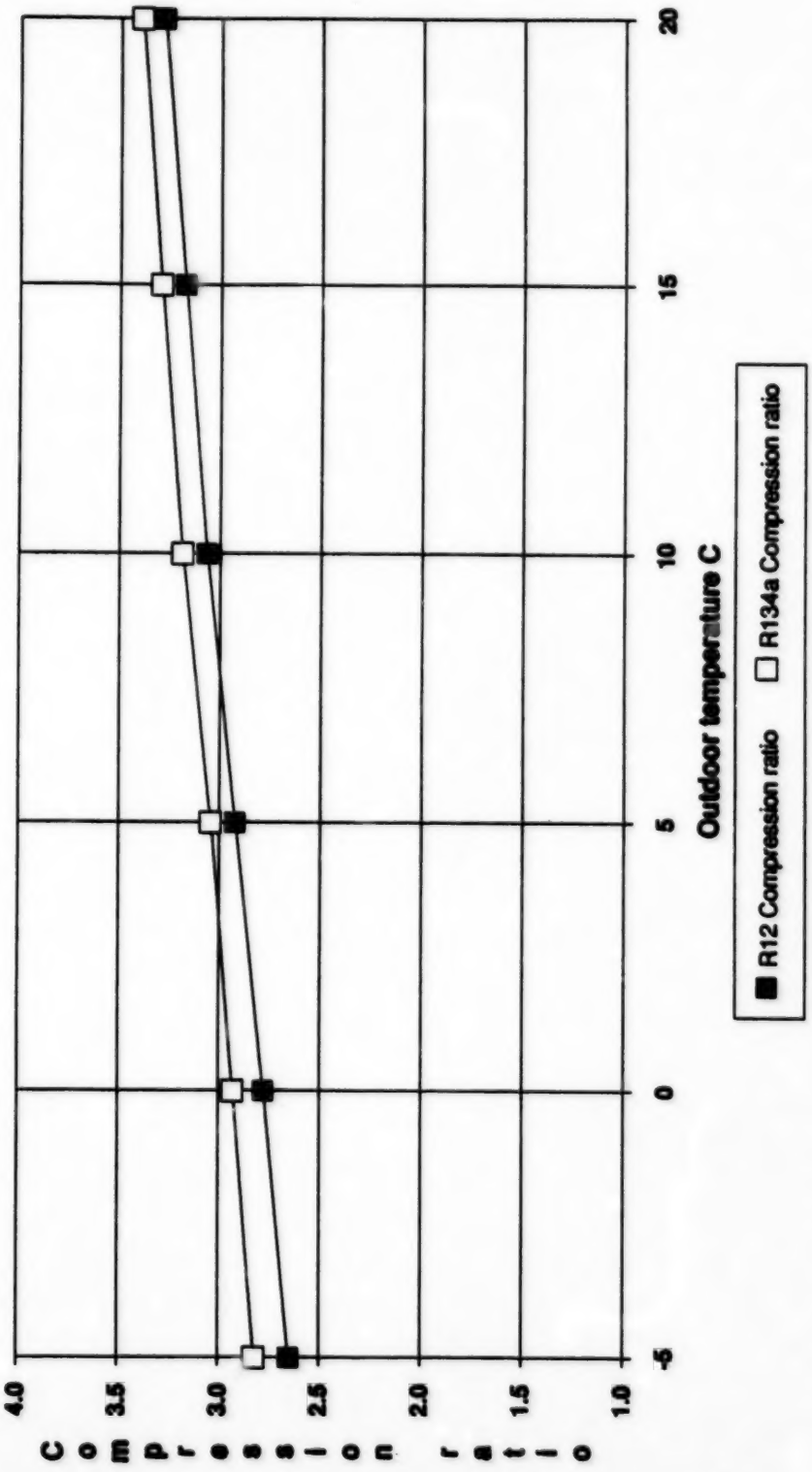

Figure 9. Comparison of R12 and R134a compression ratio

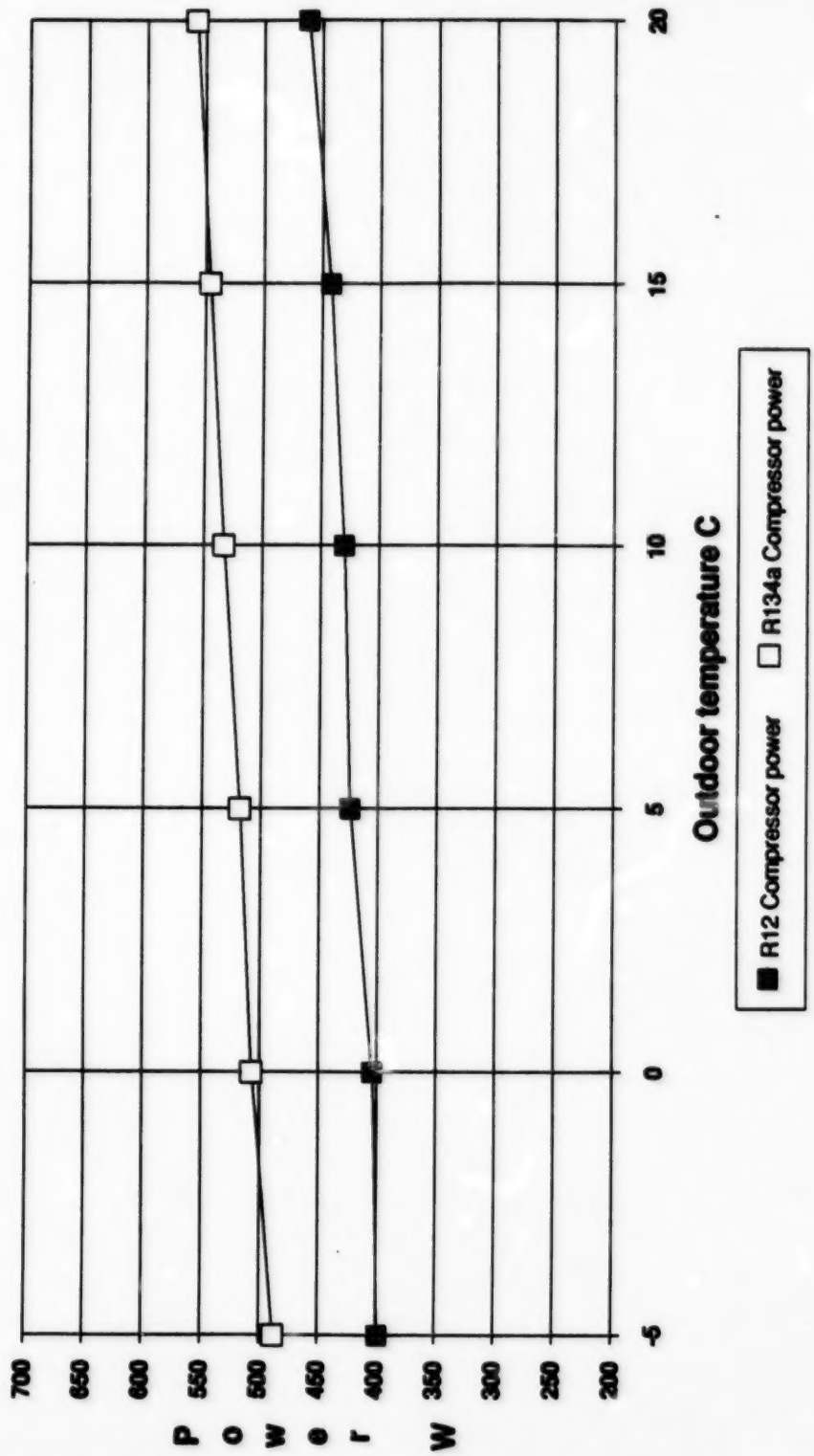

Figure 10. Comparison of R12 and R134a compressor power

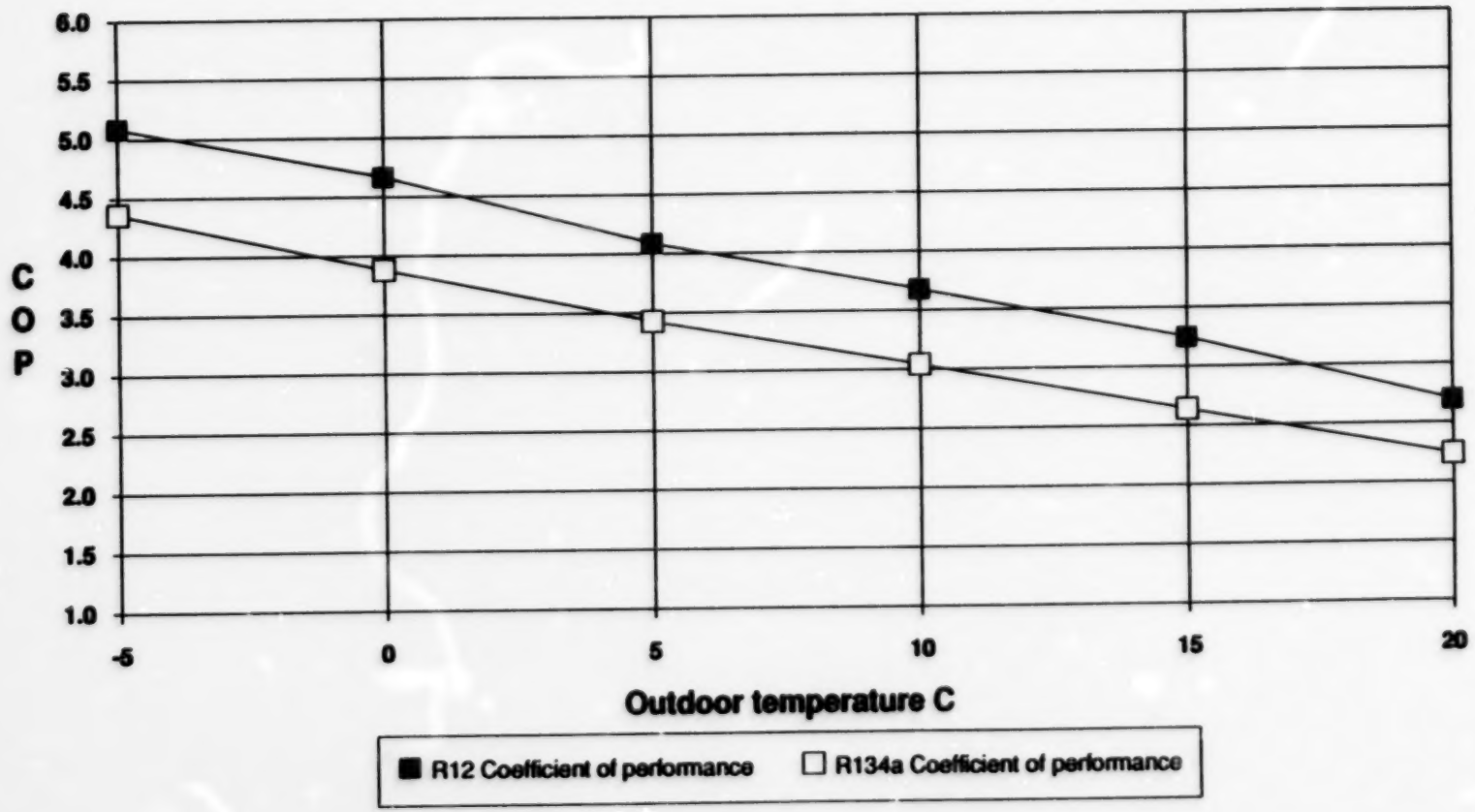

Figure 11. Comparison of R12 and R134a coefficient of performance

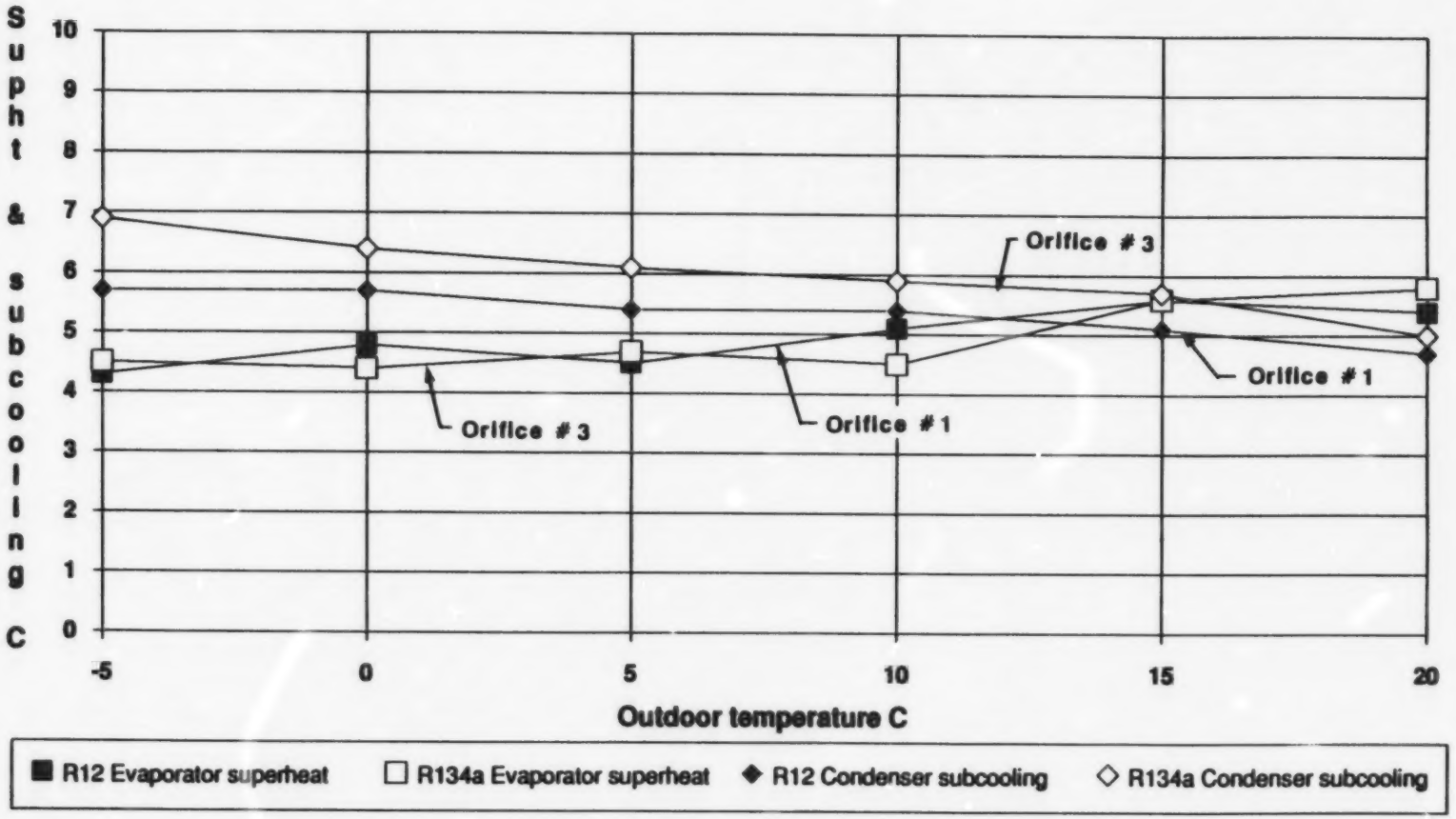

Figure 12. Comparison of R12 and R134a superheat and subcooling

Table 1.

R-12 and R-134a Test Results

	Outdoor Temperature (C)					
	-5.0	0.0	5.0	10.0	15.0	20.0
R-12 Capacity (W)	2024.0	1911.0	1735.0	1582.0	1435.0	1245.0
R-134a Capacity (W)	2135.0	1975.0	1780.0	1619.0	1441.0	1250.0
R-12 Compressor Power (W)	399.0	404.0	424.0	430.0	443.0	462.0
R-134a Compressor Power (W)	488.0	507.0	518.0	533.0	546.0	558.0
R-12 Coefficient of Performance (COP)	5.1	4.7	4.1	3.7	3.3	2.7
R-134a Coefficient of Performance (COP)	4.4	3.9	3.4	3.0	2.6	2.2
R-12 Condenser Inlet Pressure (kPa)	823.4	879.6	960.1	1038.1	1125.2	1204.6
R-134a Condenser Inlet Pressure (kPa)	923.7	985.0	1063.0	1150.0	1235.0	1335.7
R-12 Condenser Outlet Pressure (kPa)	808.2	866.0	955.0	1032.4	1118.9	1197.0
R-134a Condenser Outlet Pressure (kPa)	907.5	968.3	1054.9	1142.7	1227.7	1328.2
R-12 Condenser Pressure Drop (kPa)	15.2	13.6	5.1	5.7	6.3	7.6
R-134a Condenser Pressure Drop (kPa)	16.2	16.7	8.2	7.3	7.3	7.5
R-12 Evaporator Pressure (kPa)	310.3	316.5	328.5	338.6	354.0	366.7
R-134a Evaporator Pressure (kPa)	327.0	336.0	349.1	360.7	374.5	393.4
R-12 Compression Ratio	2.7	2.8	2.9	3.1	3.2	3.3
R-134a Compression Ratio	2.8	2.9	3.0	3.2	3.3	3.4
R-12 Evaporator Superheat (C)	4.3	4.8	4.5	5.1	5.6	5.4
R-134a Evaporator Superheat (C)	4.5	4.4	4.7	4.5	5.6	5.8
R-12 Condenser Subcooling (C)	5.7	5.7	5.4	5.4	5.1	4.7
R-134a Condenser Subcooling (C)	6.9	6.4	6.1	5.9	5.7	5.0
R-12 Compressor Discharge Temperature (C)	66.6	68.2	70.5	73.3	75.9	78.2
R-134a Compressor Discharge Temperature (C)	76.6	78.5	80.4	82.4	83.9	86.4
R-12 Discharge Superheat (C)	15.0	14.0	12.8	11.6	11.5	10.9
R-134a Discharge Superheat (C)	21.3	20.4	20.1	19.6	18.3	17.9

Table 2

Comparison of thermostatic expansion valve performance

Operating conditions: 15°C outdoor and 22°C, 40% RH indoor

Refrigerant	Thermostatic expansion valve orifice no.	Nominal R12 thermostatic Expansion valve capacity* kW	Superheat °C	Subcooling °C
12	1	1.05	5.6	5.1
134a	1	1.05	15.3	5.5
134a	2	1.76	17.1	7.7
134a	3	3.50	5.6	5.7

Operating conditions: 0°C outdoor and 22°C, 40% RH indoor

Refrigerant	Thermostatic expansion valve orifice no.	Nominal R12 thermostatic expansion valve capacity* kW	Superheat °C	Subcooling °C
12	1	1.05	4.8	5.7
134a	1	1.05	19.2	8.4
134a	2	1.76	17.3	7.8
134a	3	3.50	4.4	6.4

* Nominal capacity rated at 4.4°C evaporating and 32.2°C condensing with 5.5°C subcooling.

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4 Preliminary Test Results Comparing Heat Pump Performance of Refrigerants R-134a and R-12

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NOTES

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14 Chlorofluorocarbons; alternative refrigerant; heat pump; performance; testing; expansion valve.

SUMMARY/SOMMAIRE

15 The paper describes the performance of an exhaust air heat pump operating with a new environmentally safe refrigerant R-134a compared to the original CFC refrigerant R-12.

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